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EXPERIMENTAL RESULTS OF A DARRIEUS TYPE VERTICAL AXIS ROTOR, IN A WATER CURRENT

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Division of
Mechanical Engineering

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The author has benefited from the results provided by Nova Energy Limited, of Dartmouth, Nova Scotia, who studied extensively the Darrieus turbine, and designed and manufactured the Cornwall machine.

ABSTRACT

This report describes model tests performed on vertical axis, straight blade, Darrieus type rotors, used to extract energy from fast flowing waters.

Ducted turbines with a shroud around the rotor are investigated in configurations simulating free stream (no head) and ultra low head (low dam developing 1 to 2 m head full scale) situations.

In addition, a full scale prototype turbine installed in the St. Lawrence River at Cornwall, Ontario, as a field experiment, is described, and test results provided.

The turbine characteristics are shown in terms of efficiency, discharge coefficient and power coefficient. Efficiency of 35% has been measured.

Test results are compared with theoretical prediction algorithms developed during the course of the study.

RÉSUMÉ

Des essais sur maquette d'une roue de type Darrieus, à axe vertical et ayant ses pales droites, sont décrits. Cette roue est utilisée pour extraire l'énergie de courants d'eau rapides.

Des turbines montées dans des conduits de formes diverses sont étudiées, dans des installations au fil de l'eau (hauteur de chute nulle) et de très basses chutes (1 à 2 m à l'échelle du prototype).

De plus, un prototype de turbine installé sur le fleuve St-Laurent à Cornwall en Ontario, est décrit, et les résultats des essais sont fournis.

La turbine est caractérisée par son rendement global, son coefficient de débit, et son coefficient de puissance. Un rendement maximum de 35% a été mesuré.

Les résultats expérimentaux sont comparés aux algorithmes théoriques développés durant le cours de l'étude.

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SYMBOLS

Symbol	Meaning	Unit
A_A	Duct throat cross section area	m^2
A_E	Duct entrance cross section area	m^2
A_t	Rotor frontal area	m^2
BR	Blockage ratio - BR = A_t/A_A	-
C	Blade chord	m
C_l	Round bar drag coefficient	-
C_D	Discharge coefficient - $C_D = V_A/\sqrt{2\ g\ \Delta H}$	-
C_K	Power coefficient - $C_K = P/(.5\ \rho\ A_t\ V_A^3)$	-
g	Gravitational acceleration	m/s^2
K	Rotor drag coefficient	-
n	Number of blades	-
P	Shaft mechanical power output	W
Q	Total discharge through the duct	m^3/s
R	Rotor radius	m
S	Rotor solidity - S = n C/R	-
V_A	Average duct throat velocity = Q/A_A	m/s
V_E	Average duct entrance velocity = Q/A_E	m/s
V_0	Average upstream incoming velocity	m/s
α	Blade fixed pitch angle	o
ΔH	Total head difference across the duct or dam	m

SYMBOLS (Cont.)

Symbol	Meaning	Unit
	Turbine efficiency	-
	$\eta = \frac{P}{0.5\ \rho\ A_E\ V_0^3}$ for ducted free stream turbine	
	$\eta = \frac{P}{0.5\ \rho\ A_t\ V_0^3}$ for unducted free stream turbine	
	$\eta = \frac{P}{\rho\ g\ Q\ \Delta H}$ for a restricted flow turbine in a dam	
ρ	Specific mass of water	kg/m^3
ω	Rotor angular velocity	rad/s

EXPERIMENTAL RESULTS OF A DARRIEUS TYPE VERTICAL AXIS ROTOR, IN A WATER CURRENT

1.0 INTRODUCTION

In 1978 the National Research Council Canada initiated a study for the investigation of new devices capable of extracting energy from flowing water in rivers, or other water currents.

From various candidates, two lift devices were retained for further analysis:

Firstly, a rotor with its axis parallel to the main water flow, having two or more blades turning in a plane perpendicular to the main shaft, looked attractive because of its similarity to an autorotating helicopter rotor, and therefore provided a new application for this type of rotor.

Secondly, a rotor with its axis perpendicular to the main flow, having its blades parallel to the main shaft (a Darrieus type rotor) looked also attractive because of its simplicity (especially if the shaft is vertical), and also because this type of rotor had already showed some promising results with the vertical axis wind turbine which was being developed by NRC.

A possible application of this rotor would be to extract energy from high tidal currents such as in the Bay of Fundy. It was felt at that time that one or more large rotors (20 m in diameter, 15 m high), mounted on a vertical shaft under a floating barge, might be used instead of a conventional type turbine installed in a dam across the Bay of Fundy. This type of machine would have the advantage that the generator would be mounted on the barge, out of the water, thus simplifying the design of the whole machine. Other advantages would include:

- reduced initial capital investment since each machine can be put into service immediately, and provide early return on investment, as other machines are being constructed.

- same direction of turbine rotation, whether the tide is ebbing or flooding.

- minimal environmental effect since the water flow is not completely blocked by a dam.

- power production spread more evenly over the tidal cycle, since this type of turbine operates even at very low head differences.

All these advantages looked very attractive, but of course a lot had to be done before a definite answer could be given on the feasibility of the project.

At about the same time, the Coriolis program in the United States [Ref. 1] was investigating a horizontal axis machine that would generate power from the current of the Gulf Stream. On a much smaller scale, the Intermediate Technology Development Group, from the University of Reading in England [Ref. 2] was investigating current energy systems, with the construction of small Darrieus type rotors which were mounted on the bow of a motor boat and tested at various speeds in still water. These tests led to the installation of several pontoon supported machines on the White Nile River in Southern Sudan, Africa, which are used for irrigation pumping.

D. J. Hilton [Ref. 3] in Australia studied the theoretical aspect of the Darrieus rotor, following the works by J.H. Strickland [Ref. 4] and R.J. Templin of NRC [Ref. 5]; he then built a model which he tested by mounting it beneath a boat which moved the rotor through still water to create the necessary current.

All these hydraulic machines provided an efficiency which was comparable to that of vertical axis wind turbines. Hilton measured during the tow tests a maximum efficiency of 31% which came very close to what had been predicted. The ITDG group predicted an efficiency of 25%.

The kinetic energy available from tidal currents such as off the British or the Australian coasts have been assessed. A similar assessment for the Canadian coast and Canadian rivers [Ref. 10], showed that thousands of megawatts are potentially available, waiting for a practical device to be built to harness this potential. The National Research Council Canada then initiated preliminary studies to investigate which candidate technology would be most suitable. Most of the studies and subsequent work has been performed in association with Nova Energy Ltd. of Dartmouth, Nova Scotia as contractor to NRC.

2.0 PRELIMINARY INVESTIGATION

The first study was to choose between a vertical and a horizontal axis machine.

2.1 Model Description

Three turbine models were built:

a) A straight bladed Darrieus vertical axis rotor (diameter 81 cm, blade length 61 cm), manufactured in aluminum and mounted under a small barge. The solidity S varied from 0.15 to 0.45 by having 1, 2 or 3 blades. The blade chord C was 6 cm. This model was a scaled version of the huge machine that could be used in the Bay of Fundy tidal currents [Fig. 1 and 2].

b) A horizontal axis machine [Fig. 3], where two blades were mounted at right angles to the main shaft, providing a rotor 81 cm in diameter. This machine was closely similar to an auto-rotating helicopter rotor. It was not self-starting, but turned very quickly as soon as an initial rotating motion was provided. A right angle gear was required to drive a vertical shaft inside a watertight housing.

c) A two-bladed vertical axis machine (petal turbine) where the blades were mounted at 45° to the main shaft, and thus described a cone when in rotation.

These three models were installed in a specially built flume, 2.74 m wide, 1.3 m deep and 11 m long. The flume is capable of a maximum discharge of 1.9 m^3/s.

In all cases, the foil section for the blades was the NACA 63_4-021 basic symmetric thickness form (refer to Ref. 6 for details on this section). This selected section has characteristics judged to be well suited to the vertical axis machine. It should give low drag characteristics and high lift to drag ratio. Its maximum thickness (21% of chord length) should give high resistance to bending. Its basic hydrodynamic characteristics were derived from Abbott and Doenhoff [Ref. 6].

The output power of the three models was estimated using an improvised prony brake. It did not provide a reliable torque measuring system, but gave good relative values. It had the disadvantage that torque fluctuation could not be measured, and also that it was not possible to measure the power generated on the rising portion of the curve of power versus tip speed ratio.

2.2 Model Test Results

All three models operated well.

a) The petal turbine produced less power than the other two and therefore it was not investigated further. This can be compared to the top and bottom ends of the two foils of the NRC Darrieus "egg-beater" style wind turbine, which, due to their tilt relative to the wind, contribute much less torque than the sections of the blades close to the equator of the rotor. The petal turbine will turn in both horizontal and vertical axis configurations, and always in the same sense of rotation.

b) The horizontal axis machine produced power comparable to the vertical axis machine (straight blades), but it turned two to three times faster. This could be an advantage since a smaller speed increasing gear box on large prototypes would be required. But on the other hand, the right angle drive required to bring the shaft to the generator mounted on the barge, did not seem practical for underwater use. It was also noted that the blades would bend backwards due to the high lift loads on the foils, and excessive bending stresses would be found at the root of the blades in the case of large prototypes. Since the machine was not self-starting, and since at the time, it was not felt desirable to have the generator under water, the horizontal axis machine was not tested further.

c) The straight bladed vertical axis Darrieus rotor was initially tried in two configurations:

First, a set of straight walls was installed to narrow the flume to a width of only 1.8 m. This had the effect of increasing the velocity through the rotor, but also increased the ratio of the rotor frontal area to the flume cross section (blockage ratio BR). The barge was rigidly mounted between the two walls, under a bridge support.

A maximum efficiency $\eta = 0.47$ was obtained with a three-bladed rotor (solidity 0.45). The efficiency had been defined as

$$\eta = \frac{P}{1/2\ \rho\ A_t\ V_o^3} \qquad (1)$$

where P = shaft power output
A_t = frontal area of rotor
V_o = average incoming velocity upstream of the turbine

so that the power could be compared with the kinetic energy which approached the rotor frontal area.

Next, in order to be able to install the barge complete with its mooring, the full width of the flume (2.74 m) was made available, with the barge free to float on the surface, held in position by four mooring lines. The velocity of the water was much smaller of course, but the blockage ratio decreased to 0.2. Due to the presence of the flume walls, this set-up still did not represent a free stream situation (BR = 0), but it was the closest that could be achieved. During the tests, the barge showed some gentle oscillatory motion, in phase with the rotation of the rotor. The measured efficiency was of the order of 0.38, a significant drop compared to the tests with the 1.8 m wide flume.

It was felt that this efficiency drop was mainly due to the blockage effect. In an unbounded flow field (infinitely wide channel), the Betz effect applies, whereby the flow is able to go around as well as through the rotor, and continue downstream. Betz predicted that only 59.3% of the approaching kinetic energy could be extracted from the flow by a perfectly efficient machine. But when the blockage ratio is increased by narrowing the channel walls, the fluid stream lines are forced through the rotor, reducing the Betz effect, and increasing the power output. It was then decided to study more extensively this blockage effect by installing around the rotor a series of shrouds, or ducts, of various shapes that would enhance the flow through the rotor.

3.0 RESTRICTED FLOW TESTS

Due to the limited width of the flume, it was not possible to design a model of the rotor and its associated ducts of sufficiently small size, yet giving reasonable power output, to simulate a totally

free stream situation. Ideally, the flume blockage by the duct should not exceed 10% of the flume cross-section area. If this had been the case, the rotor blades would have been much too small to manufacture accurately, and the blade Reynolds number would also have been too low. It was then decided to close the flume cross-section completely by installing the rotor in an opening specially designed for the installation of the ducts. In this case, the configuration was not a free stream machine but a turbine located in a barrage or low dam, with a head developed across it. This enabled the study of the effect of the blockage of the rotor relative to the cross-sectional area of the duct. It was anticipated of course that since the power output was proportional to the cube of the velocity, and since all the water now was forced through the rotor, a larger power output would be obtained.

3.1 Experimental Set-up

Five different shapes of duct were tried, as shown in Figures 4 to 8, each attempting to enhance the rotor performance.

Fig. 4. The caissons were intended to guide the flow radially onto the rotor, so that the blade angle of attack would be more uniform, as the blade passes in front of the upstream flow.

Fig. 5. The flat wall was intended to increase the Betz coefficient by confining the stream lines close to the rotor, not allowing them to diverge as the water approached the rotor.

A modified version of the flat wall duct was also tried. A low bump (5 cm high) was fastened vertically, in the centre of the wall. The intention was to locally change the direction of the flow in an area where the angle of attack was close to zero. This would minimize the time during which the blade did not produce any torque (near rotational angles of 0 and 180°, adjacent to the guide walls).

Fig. 6. On the exit side of the duct, the flow re-entered an area of high pressure and low water velocity. In order to minimize the head loss due to this abrupt expansion of the duct, a set of vanes was installed between the rotor and the caissons, thus providing a gap to re-energize the boundary layer on the downstream side of the caissons, to retard flow separation somewhat from the caissons.

It was noted at this point that a long draft tube might have been preferable, but the idea was to keep the duct short and symmetrical, so that the same configuration could be used in a tidal situation with reversing flow.

Fig. 7. In order to direct the flow onto the rotor, and then divert it away, a set of four guide vanes was installed. The vane angle could be changed from 45° to 60°, relative to the flow channel direction.

In all the above tests, the top and bottom surfaces of the duct were as shown in Figure 8. The foil section was the same as used in the preliminary tests: NACA 63$_4$ - 021. The three-blade rotor, 61 cm high and 81 cm in diameter, was made of aluminum [Fig. 9]. The arms supporting the blades were machined to approximate a foil profile with a round leading edge and a thin trailing edge to minimize the drag. No attempt was made to shape them accurately to any foil section. The torque was measured with a Lebow rotary transformer torque sensor which also provided the RPM by means of a magnetic pickup. The speed of rotation of the rotor was set by an induction motor-generator coupled to the main shaft through a variable speed belt-drive. A continuous range of RPM between 50 and 300 was thus available.

The water levels upstream and downstream of the turbine were measured with two capacitive probes located in stabilizing tanks. The total discharge Q of the flume was obtained from a calibrated sluice gate located at the end of the flume.

3.2 Experimental Results

Several series of tests were conducted between 1980 and 1982. Full details can be obtained from Reference 7. Only the most important results are explained below.

The turbine characteristics are now defined by:

(1) the discharge coefficient

$$C_D = \frac{V_A}{\sqrt{2\ g\ \Delta H}}$$

where V_A is the average velocity at the throat of the duct and ΔH is the total head difference (static and dynamic) across the duct; and

(2) the power coefficient

$$C_K = \frac{P}{1/2\ \rho\ A_t\ V_A^3}$$

where P is the turbine shaft power output and A_t the frontal area of the rotor.

C_D represents the ability of the water to go through the duct and rotor. It is expected that C_D will decrease as the RPM increases. C_D has been represented as a function of the dimensionless tip speed ratio

$$\frac{R\ \omega}{V_A}$$

where R is the radius of the rotor and ω the angular velocity.

C_K represents the ability to extract power from the flow, and has also been plotted as a function of the tip speed ratio. The total efficiency of the turbine is now defined as

$$\eta = \frac{P}{\rho g Q \Delta H} = C_K \, BR \, C_D^2$$

where Q is the discharge through the duct. This new definition is now required because the turbine experiences not only incoming kinetic energy, but also a head drop across it.

The effect of the blockage ratio, BR, is shown in Figure 10. The discharge coefficient decreases as BR increases because of the restriction caused by the close walls, whereas both C_K and η increase significantly. This can be explained by the fact that less water escapes through the gap between the rotor and the duct, and thus the Betz coefficient (if one could define such a coefficient in a restricted flow) increases.

Figure 11 shows comparative results between three different duct configurations. The caisson and vane produces the best efficiency, due mainly to a higher discharge coefficient. The gap between the caisson and the vane seems to help the flow expand in the duct exit. The caisson alone produces a broad efficiency curve which is more desirable than a peaked curve. The modified walls produce both C_K and η curves which are higher than in the case of the straight wall, for the same blockage ratio.

As a matter of comparison, Figure 12 shows the C_D, C_K and η coefficients obtained when the duct sidewalls are removed entirely. Table 1 shows comparative power output for the same model, with and without the caisson duct.

In all cases the tip speed ratio for maximum efficiency is smaller than that corresponding to the maximum power coefficient. This is because the discharge coefficient drops very quickly as the tip speed ratio increases. This drop in C_D causes less flow to go through the rotor, and more flow through the rotor to caisson gap.

It should be noted that the efficiency obtained is still rather small (of the order of 30%). This is probably due to the shape of the duct which produces a very abrupt expansion, creating large head losses as the flow decelerates to join the tailrace of the barrage.

On the other end, even with such a low efficiency, the net shaft power is much higher, when the turbine is installed in a barrage structure, than when all the duct shapes are removed (shown in Table 1), since all the flow and a greater head, thus greater velocity, is forced to act on the rotor.

On large prototype machines, installed in a low head dam, with the efficiency being rewritten as

$$\eta = C_K \, BR \, C_D^2$$

special precautions will have to be taken in the design of the duct (or draft tube) in order to increase C_D and the blockage ratio. The increase of C_K can be obtained by choosing better foil sections, and reducing the blade tip losses and the interference drag between the blades and the support arms.

CONFIGURATION / PARAMETERS	CAISSON	ALL DUCTS REMOVED TOP AND BOTTOM STRUCTURE STILL IN PLACE (FIGURE 8)
Blockage ratio	0.8	0.26
Water elevation difference (cm)	14.8	1.6
Total flume discharge (m^3/s) = Q	0.885	1.92
Approaching velocity (cm/s)	24.9	72.6
Duct velocity (cm/s)	139.1	99.2
Shaft power P (watts)	432	65
RPM	110	60
Efficiency $\eta = \frac{P}{\rho g Q \Delta H}$	0.34	0.22

Table 1 Comparative Power Output (model test results)

4.0 INVESTIGATION OF THE OPTIMUM BLADE ANGLE OF INCIDENCE

In the spring of 1982, a series of tests was conducted in order to investigate the effect of the blade fixed angle of incidence on the power output, [Fig.13]. A set of small wedges was prepared and inserted between the blades and the support arms.

Since these wedges would introduce an extra drag load due to their interference, it was decided to prepare a set of flat spacers. These spacers would reproduce the drag of the wedges, without changing the angle of incidence of the blade. The model used was the same rotor

as before (diameter = 81 cm, height = 61 cm, solidity 0.45), mounted between the two semi-circular caissons (BR = 0.8). The results of the tests are shown in Figures 14 and 15 for positive (outward) and negative (inward) angle of incidence [Fig. 13]. It can be seen that the interference drag alone causes the power coefficient to drop by about 5%.

The original blade angle was set at 0° when the angle was defined at the half chord location. But, to be consistent with generally accepted aerodynamic theories, this translated to a positive angle of incidence of approximately 2°, based on the quarter chord distance from the leading edge.

It appears that for the particular set of blades and solidity, the optimum angle of incidence is about 2°. This is consistent with the conclusions of Stacey and Musgrove [Ref. 8] and of Wilson and McKie [Ref. 9], in the case of Darrieus wind turbines.

5.0 TURBINE OUTPUT POWER THEORETICAL PREDICTION PROGRAM

In 1981, a contract was awarded to Nova Energy Ltd. of Dartmouth, N.S. to prepare a computer program predicting the performance of the rotor, when mounted in a duct. Complete details of the program can be found in Reference 7.

The theory used was that of Templin [Ref. 5], which was modified to account for the presence of the duct. Two stream tubes were used. One going through the cross-section of the rotor itself, and the other incorporating only the gap between the rotor and the duct. It was felt that since the power output was so dependent on the blockage ratio, the flow not going through the rotor had to be identified separately.

In the derivation of the program the following major difficulties were encountered:

a) Exact foil characteristics unknown for low and high Reynolds number. The final drag and lift coefficient derived from Abbott and Doenhoff [Ref. 6], were extended according to Templin [Ref. 5], and modified until the program output matched more or less the model test results.

b) Rotor support arm drag. This was measured separately on the rotor, with the blades removed.

c) Duct losses. Estimated using model test results.

d) Inability of the program to represent exactly the flow velocity vector as it entered and left the duct.

After several trial runs, satisfactory results were obtained [Fig.16] for a given set of circumstances. If the circumstances are changed (Reynolds number, different duct shapes), it is not known if

the program will provide similar results, and considerable study still remains to be done. It was felt at this stage, however, that it would be preferable to try out the Darrieus rotor on a large scale prototype which would provide valuable field experience, rather than trying to refine the prediction program.

6.0 FIRST FIELD TEST - FREE STREAM TURBINE WITH DUCTED ROTOR

After having spent several years on model testing and theoretical analysis, the need for a field test was becoming apparent in order to try out at a larger scale the results of the laboratory studies. It was also time to get hands-on experience with heavy hardware such as bearings, gear boxes, generators, corrosion-resistant materials, mooring system, etc. A prototype turbine was built and installed in 1982.

6.1 Turbine Description

The type of turbine chosen was a free stream vertical axis unit, similar to the one used during the preliminary barge test in 1979-1980. The three bladed rotor would be suspended under a floating structure and moored in the river. In order to try to improve the efficiency it was decided to install a duct around the rotor. The machine would also be symmetrical so that it could be redeployed, in subsequent tests, in tidal areas. Nova Energy Ltd. was given a contract to design and build the machine.

The duct has the same general shape as the "caisson and vane" test [Fig. 17], but the "caisson" has a much larger radius in an attempt to decrease the very large head losses due to the expansion of the flow on the exit side. It is equipped also with vanes providing a gap for the water to re-energize the boundary layer. This configuration had provided the maximum efficiency during the model tests. The ratio of the duct throat area to the duct entrance area is 0.53. The duct is composed of several removable sections. Each section is made of aluminum sheet bent and riveted to heavier aluminum strength members. The sections are then bolted to the main steel frame on which are mounted the gear box and generator [Fig. 18].

The gear box serves two purposes. Firstly, its output shaft turns at 22.33 times the rotor speed by means of a double helical gear-train; and secondly, it supports the main vertical rotor shaft. This is accomplished using an off-the-shelf 41 kW rated Eurodrive agitator, bolted to the top of the main steel frame. This arrangement eliminates the bottom bearing, and any coupling between the main shaft and the speed increasing gear box. The generator, a 4-pole induction motor rated at 22 kW, is coupled to the gear box by two belts. The rpm of the rotor is varied by changing the belt pulley ratios.

In order to keep the drive train simple, there is no torque sensor in the system. However, the generator and various belt sets were calibrated in the laboratory with a 30 kW DC motor driving the generator. Both the mechanical input and electrical output powers, to and

from the generator, were measured, thus providing the efficiency of the system at various loads. The efficiency of the gear box was not measured accurately but rough estimates were obtained using a torque wrench on the low speed shaft.

The rotor is made of three extruded aluminum blades, with the NACA 634-021 profile. The blades (1.5 m long with a 0.2 m chord) are supported by two arms of the same extrusion as the blades. The arms are bolted to the blades. The hub assembly is simplified since the low speed shaft of the gear box is very long, so that all six arms can be bolted to several "sandwich" plates, mounted and keyed to the shaft. The rotor is 2.4 m in diameter giving a solidity of 0.5. This is probably not the optimum solidity expected to be around 0.35. It is high compared to the usual Darrieus wind turbines, but it is thought that the same blade extrusion can be used on even larger water turbines, thus obtaining a smaller solidity closer to the optimum.

The whole system assembly (weighing approximately five tonnes) is supported by two fiberglass covered, foam filled, pontoons, 0.9 m in diameter and 6.1 m long [Fig. 19 and 20]. The generator, gearbox, pulleys and belts (located mainly below the water surface) are enclosed in a watertight superstructure equipped with a hinged top cover providing access to the inside equipment.

6.2 Site Description

The site had to have the following characteristics:

- high water velocity (minimum 1.5 m/s).
- sufficient depth.
- access road.
- 600 v 3-phase electrical power available nearby.
- no interference with local boat traffic.

A site at Cornwall, Ontario, on the St. Lawrence River, has all these characteristics. An overhead 12,000 volt power line crosses the river at this location. The St. Lawrence Power Company, very efficiently, supplied three transformers providing 600 volts, available 30 m from the turbine. The Company also supplied the power meters for measuring the power to and from the turbine, independently.

The site is located about 2 km downstream of a power dam, which stops all floating debris, and therefore there was no need to install a trash rack in front of the turbine.

During the original river current water survey, it was found that the surface velocity was close to 2 m/s. However, further measurements showed that there was a steep vertical velocity gradient,

and that the duct would experience an average velocity of only 1.5 m/s. The effect of the power dam discharging various amounts of water depending on the time of day was felt noticeably at the site. The velocity measured varied significantly, which made the power measurements difficult to interpret.

The access road is located on the river bank, approximately 10 m above the water, and 10 m away from it. It was impossible to deploy the turbine directly from that location with a crane. A suitable launching area was found about 3 km downstream, at a wharf with sufficient water depth and direct access for the low-bed trailer and a crane. Between the wharf and the desired site, the river narrows, creating very high velocities (2 to 2.5 m/s). This location would have been ideal for the turbine installation, but would have affected the local boat traffic, and furthermore had no 600 V power nearby.

6.3 Turbine Installation

The ducted turbine was assembled on the wharf in Cornwall and loaded onto a barge equipped with a crawler crane. Then the barge was towed upstream with a 400 hp tug. It soon appeared that the tug would not be powerful enough to tow the barge in the narrows because of the high water current. A second tug (265 hp) was called in to assist the first one, and the two tugs successfully reached the installation site. Once in position, the crawler crane lifted the turbine and lowered it in the water. The unit was held in position by two temporary mooring lines preventing the turbine from drifting with the current. Once the permanent mooring lines were installed it was realized that one of the anchors was dragging on the bottom, allowing the turbine to move laterally into an area of very low currents close to shore. It was decided to redesign the mooring configuration, and not to use anchors but to moor the turbine to an old bridge pier, and to a very heavy piece of cast iron which happened to be embedded in the river bottom.

Moving the turbine away from the shore to its new moorings took three full days of site preparation and exhausting manual effort. One only has to feel the drag force of a 2 m/s current on a diver to appreciate the energy in flowing water. The large powerful tugboats could not manoeuvre easily in the currents, so a small outboard motor craft, which of course could not provide the pulling force required, was used for passing a line from one place to another. The actual pulling was done by hand with 5 tonne "come-along" cable pullers. When the turbine was in its final position, two overhead electrical cables (one for the power output, the other to carry the electrical control signals) were installed by the St. Lawrence Power Company between the turbine and the shore [Fig.21].

6.4 Prediction of the Ducted Free Stream Turbine Output Characteristics

The model tests described in Section 3 were performed with the rotor installed in a barrage where the water velocity was known. In a ducted free stream situation however, the amount of water going through

the rotor is unknown. It is also very difficult to define a duct discharge coefficient C_D since the head across the duct is very small, and the flow, on the exit side, does not present a clearly defined velocity distribution or free surface. Therefore tests were required to assess the effect of the duct and rotor on the flow pattern through the duct.

As mentioned earlier, due to the limited width of the flume, it was not practical to model the complete ducted turbine in a true free stream situation (i.e. negligible blockage effect due to the presence of the model in the flume). It was decided to prepare a model of the duct only, at a scale of 7.5, without the rotor [Fig. 22]. The rotor was simulated by a set of vertical round bars, equally spaced at the throat of the duct. These bars presented a restriction to the flow similar to the rotor and caused a total pressure drop across them.

Velocity measurements of the flow were made at the duct throat, and in planes located one turbine diameter behind the bars, and four turbine diameters in front of the bars. Water levels were measured upstream and immediately downstream of the duct.

A relationship between the bar spacing and the RPM of the simulated rotor was obtained by equating the drag force on the bars to the drag force on the rotor.

The drag force on the bars was derived from the expression:

$$F_M = A_M \rho C_1 \frac{V_A^2}{2}$$

with F_M = drag force on the bars in the model

A_M = cross section area of the throat of the duct model

V_A = average velocity through the duct throat

C_1 = drag coefficient of bars alone (after substracting the duct only losses).

The drag force on the rotor was similarly derived from the expression:

$$F_R = A_t K \frac{1}{2} \rho V_A^2$$

where F_R = drag force on the turbine rotor

A_t = frontal area of the rotor

K = rotor drag coefficient.

By equating F_R and F_M, a relationship between K and C_1 was obtained. Since K was defined in terms of the tip speed ratio, and C_1 in terms of the bar spacing, a similar relationship could be derived between the tip speed ratio and the bar spacing.

For this transformation, the numerical values of K (Figure 24) were obtained from the theoretical prediction program derived by Nova Energy Ltd. (see section 5). This program had only been validated for a duct having straight parallel walls as shown on Figure 5. It was felt that such a prediction would nevertheless be acceptable in this first attempt to predict the efficiency of a free stream ducted turbine.

The numerical values of C_1 were obtained by measuring for various bar spacings, the total head drop accross the bars, and the average flow velocity V_A through the duct.

The turbine efficiency

$$\eta = \frac{P}{1/2\ \rho\ A_E\ V_0^3}$$

relative to the kinetic energy approaching the total frontal area A_E of the duct can be rewritten as:

$$\eta = C_K \left(\frac{V_A}{V_0}\right)^3 \frac{A_t}{A_E}$$

where C_K is the power coefficient defined in terms of the rotor cross section area A_t and the throat velocity V_A.

In order to calculate η, the values of C_K were used as measured during the restricted flow model test in the caisson configuration, since this was the duct shape that most nearly approached the shape of the free stream turbine duct; V_A and V_0 were measured on the duct model test for various bar spacings [Fig. 23] and then related to the corresponding tip speed ratios.

The results of the prediction are presented in Figures 25 and 26 relative to the dimensionless number $R\omega/V_0$. It is to be noted that these turbine characteristics apply only for the particular duct shape shown on Figure 17 which has a capture ratio (area of duct entrance to throat area) of 1.88.

6.5 Different Approach for Predicting the Ducted Free Stream Turbine Characteristics

In order to predict the characteristics of such a turbine, Davis made the following assumptions [Ref. 7]:

a) The static pressure in the duct exit vertical plane was the same as the upstream static pressure.

b) The flow velocity downstream of the turbine was the same as the incoming upstream velocity.

c) The discharge coefficient C_D could be represented by an empirical curve showing, for zero RPM, a value 50% higher than that measured during the caisson tests, and for high RPM, values approaching those measured.

With these assumptions, it was possible to define the throat velocity V_A, in terms of C_D, V_0, and the duct geometry. The efficiency η, as defined in Section 6.4, could then be rewritten explicitly in terms of C_K, C_D and the turbine geometry. The predicted values of η and V_A/V_0 based on Davis' assumptions are shown in Figures 25 and 26.

6.6 Test Results of the Prototype Ducted Free Stream Turbine

6.6.1 Upstream velocity measurement

The upstream flow velocity was measured from a small boat using a Marsh-McBirney electromagnetic current meter mounted at the end of a rigid rod. In the course of the experiment, two reference sets of current readings were taken, at various depths, for two different total river discharges as recorded at the upstream dam. These current measurements were taken approximately 15 m upstream of the turbine and therefore did not represent the true incoming current speed just in front of the turbine. This could have been of no major significance if the river bottom had been almost flat, but in this case, the bottom presented a ridge, in front of the machine, affecting the velocity of the water along a stream line. It was not possible, because of the mooring arrangement, to come closer to the turbine with the boat.

The vertical velocity profile presented a steep gradient (2 m/s at the surface, 0.6 m/s at 3 m depth). The velocity value retained was the simple average of the profile, calculated over the height of the duct. An average based on the cube of the profile could be defined to represent the cube relationship between P and V_0. Another average based on the square of the profile (to represent the kinetic energy per unit mass) times the mass flow (to represent the discharge) could also be defined. But considering the fact that the various layers of water through the turbine did not stay parallel, resulting in a large mixing, it was thought that a simple average would be acceptable, at the expense of a scatter of the data points.

Because of the fluctuating river flow, and since it was not possible to take a current survey each time a power measurement was made, the corresponding velocity V_0 was calculated using a linear regression between the two reference sets of current readings and the river discharge recorded at the dam. Values of V_0 between 1.33 and 1.56 m/s were thus calculated in the data analysis.

The inaccuracy of this method, and the steep vertical velocity gradient explain the scattering of the data points on the turbine characteristic curves.

6.6.2 Duct throat velocity

The current meter was installed in front of the duct entrance, and moved in a grid pattern in order to obtain a set of 25 measurements. The average of each set gave the duct entrance velocity, V_E. The average duct throat velocity V_A was then calculated by continuity, $V_A = 1.88\ V_E$. It was noted that large gradients were observed in the duct entrance set of measurements (1.17 m/s at the top, 0.48 m/s at the bottom, for example). It was also noted that, because of the wide rotor hub, the stream lines were inclined downwards giving large velocities at the bottom of the duct exit, and almost no current at the top. The flow behind the hub was also found very small.

Because of this mixing of the water, when going through the duct, no attempt was made to analyse the flow in different horizontal layers even though each layer would experience a different tip speed ratio.

6.6.3 Shaft power measurements

To obtain the shaft power output produced by the rotor, the shore power measurements of the generator were corrected for the efficiency of the generator and the mechanical losses in the belts and the gear box. Because of the unexpected low current velocity, the maximum efficiency of the drive train (including generator) was only 48%. Power measurements were made every time the pulley ratio on the turbine was changed and were related to the corresponding total river discharge recorded at the dam.

The maximum power produced between two visits at the site was also recorded and was later related to the maximum river discharge which occurred during the same time period.

6.6.4 Turbine characteristics

The data points are plotted on Figures 25 and 26 in terms of the turbine efficiency (relative to the total incoming kinetic energy across the duct frontal area) and of the velocity ratio V_A/V_0.

This last ratio shows a definite increase of the velocity through the turbine because of the duct effect, (between 1.0 and 1.5). But this increase is much less than the aperture ratio $A_E/A_A = 1.88$, which indicates that the flow slows down just before entering the duct. The scatter of the data points have been explained earlier by the difficulty of obtaining an accurate value for the reference velocity V_0.

The main characteristic of Figures 25 and 26 is that the points show a smaller RPM than expected. If the points were shifted to the right by about half a "tip speed ratio", they would fit fairly well

the predicted values. Davis suggests that this phenomenon can be explained by the higher effective solidity of the rotor. Each blade is supported by two arms having the same foil section which in fact tends to double the effective solidity. Templin [Ref. 5] has shown that such an increase causes the power curves to shift towards smaller tip speed ratios.

6.6.5 Turbine behaviour

Generally speaking, the turbine was not self starting except if, at the time when the brake was released, the blades were located in a certain position.

The maximum turbine shaft power output was 5.2 kW at 34 RPM in a 1.52 m/s current. Because of the very low efficiency of the generator and gear box, the shore power was only 2.5 kW which is very small compared to the cost of the turbine installation.

The whole duct and pontoon assembly did not show any appreciable vibration (it weighed approximately five tonnes). There was some slight yaw and pitch motion, due to the fluctuation of the river current. The unit ran for a total of 1,150 hours. The electrical safety systems functioned very well, with the unit found automatically turned off twice on arrival of the maintenance staff. This was due to a voltage fault on the main distribution grid (probably caused by lightning). There was no provision on the system for an automatic restart after such a fault.

The turbine was removed from the water at the end of the summer, using the same heavy and expensive equipment (barge with onboard crane, tugboat, shore crane), because it was felt that ice formation around the pontoons, the mooring lines and the guy lines, would probably cause too much current drag, endangering the shore electrical poles and the electrical cables. Access to the unit would also have been very difficult with a small boat even if one could launch such a craft on icy ramps.

7.0 SECOND FIELD TEST - FREE STREAM TURBINE WITH DUCT REMOVED

7.1 Turbine Installation

In the spring of 1983, in order to assess the effect of the duct on the performance of the machine, the same unit was installed at exactly the same location at Cornwall, Ontario on the St. Lawrence River. But this time, the duct around the rotor was removed leaving only the top forward and aft parts of the fairing [Fig. 27 and 28]. The generator was replaced by a smaller one (3.75 kW) in order to improve the efficiency of the mechanical drive train.

Because of the smaller draft of the machine, it was this time possible to tow the floating turbine from the launching site without the use of the barge. The tow was made using the 265 hp tugboat. When the tug arrived in the high current area of the narrow section of the river

it was discovered that she could not proceed because the turbine pontoons began to dive beneath the surface thus creating a large resistance. Since the larger tug was not available at this time it was decided to remove the turbine from the water, dismantle and load it onto a low bed float and transfer it to a second launching area upstream of the narrows. The unit was then reassembled, lowered into the water with a 60 tonne crane (required because of the long reach in order to be in deep water) and successfully towed to the site from there.

7.2 Turbine Output Characteristics

The no-duct turbine unit was equipped with a platform extending forward so that upstream velocity measurements could be taken in a section located approximately 1.6 rotor diameters upstream of the rotor shaft. An electromagnetic Marsh-McBirney field current meter was used in a grid of twelve individual points. The velocity V_0 was calculated as the average of these twelve points.

An incoming current profile measurement was performed for each pulley ratio and each power measurement thus eliminating the problem of the river flow fluctuation.

The Marsh-McBirney current meter was later calibrated on a towing carriage in the laboratory. The new induction generator and associated pulleys and belts were also calibrated in the laboratory.

The results of the measurements are plotted in Figure 29 in terms of the efficiency

$$\eta = \frac{P}{1/2\ \rho\ A_t\ V_0^3}$$

and the tip speed ratio $R\omega/V_0$. It is to be noted that the frontal area of the machine with its duct removed is A_t. Figure 29 also shows the best fitting curve for the data points obtained the previous year with the ducted turbine.

Because of the small rotating inertia of the driven system (shaft, gear box, belts, generator), the hydrofoil produced torque ripple gave a power ripple as seen on Figure 30. The ducted turbine produced a peak to peak power ripple of about 12% of the average power, whereas the unducted machine showed a 40% power ripple. This could be explained by the fact that in the ducted experiment, the generator was loaded at only 20% of rated power compared to 72% for the unducted experiment. We believe that, more likely, because of the radially shaped velocity field in the duct, each blade experienced a torque which was more uniform and therefore the ripple was minimized.

The large variation of the mean power output of the ducted turbine probably corresponds to the slow yaw motion of the unit, which allows slow variation of the amount of water passing through the duct. For the non ducted turbine, however, it is insensitive to changes in flow direction.

7.3 Turbine Behaviour

The maximum shaft power output was 4 kW at 35 RPM in a 1.9 m/s current. (Because of the early summer installation, the current was higher than in the test of the ducted turbine late in the summer the previous year.) The maximum mechanical efficiency of the drive shaft assembly (including generator) was 61% which provided 2.5 kW electrical power on shore.

The unit ran for a total of 1,189 hours. Only once was it found shut off due to a voltage fault on the distribution grid (lightning).

During this series of tests, the whole floating unit experienced a combination of pitch and roll motions of several inches in amplitude, at a rate of exactly three times the RPM, (i.e. 1 per blade). Because these oscillations could be felt at all RPM tested (including run away speed), we believe that they were not due to stalling of the blades but simply to the torque ripple. These oscillations are the same as those observed during the barge model tests described in Section 2.2. The fact that these oscillations were not felt during the previous tests with a ducted rotor, is probably due to the much heavier added mass inertia and keel-like drag of the machine which damped out the vibrations, and also to the smaller amplitude of the torque ripple, as seen in Section 7.2.

8.0 CONCLUSIONS

These series of tests have demonstrated that the kinetic energy of fast water currents can be extracted and converted into mechanical energy by using a Darrieus type vertical axis hydraulic turbine - VAHT. The efficiency of the machine remains very low, of the order of 35%, although it is felt that a few percentage points may be gained by optimizing the solidity and choosing better foil sections.

In the no-head application (free stream) a VAHT of the same type as at Cornwall is not cost effective due to the high cost of the installation equipment such as cranes, tug boats and mooring arrangements. Light and easy-to-install VAHTS, providing a very small amount of power (a few kW) could find an application in areas where diesel fuel is the main but expensive source of power. Careful site selection and judicious choice of equipment could make the small VAHT a successful energy converter.

In the ultra low head application (less than 2 m), where a low dam already exists, the VAHT is predicted to be competitive compared to low head propeller turbines, however this has yet to be proven. Because of the high drag load on the rotor, a shaft bottom bearing mounted on a base plate is required. Two side walls properly shaped around the rotor complete the flow confining duct and form the base structure that can be installed in an opening in the dam, or in front or behind of a flow regulating gate.

Such a prototype unit is now being tested in the East River near Sheet Harbour in Nova Scotia. It is expected to produce 77 kW using 15 m^3/s at a 1.5 m head.

In cases when the dam does not already exist, it is felt that, if backwater considerations permit, and where water is to be conserved, the erection of a higher dam, equipped with a more efficient conventional low head propeller turbine, would be preferable.

In the tidal application, large open VAHTs requiring only the few submerged hydrofoils (no duct) may be cost effective. Mechanical problems such as blade strength, speed increasing gears, bearings and installation methods, still have to be assessed. However, such large moored or bed-founded VAHTs may be one way of extracting energy from tidal currents without building complete dams across the entire basin.

One main problem area under investigation is the torsional vibration of the rotor due to the large torque pulses as observed during the Cornwall experiment and also during the preliminary tests of the Sheet Harbour VAHT. The long term effect of these vibrations, on the drive train and bearings is to be determined. As well, the need for a durable underwater bearing has already been recognized, and a design to encourage self-lubrication by water is under consideration.

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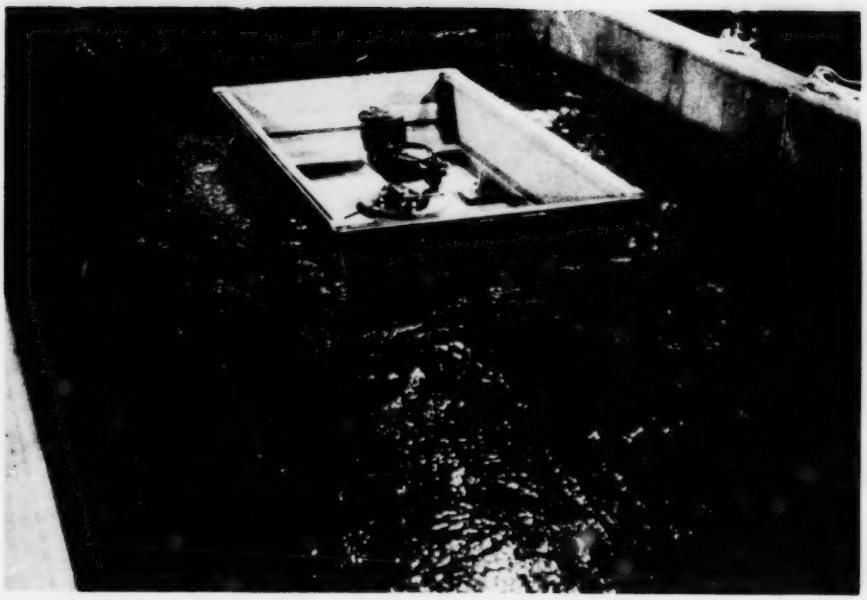


Figure 1 Floating barge model during the preliminary tests

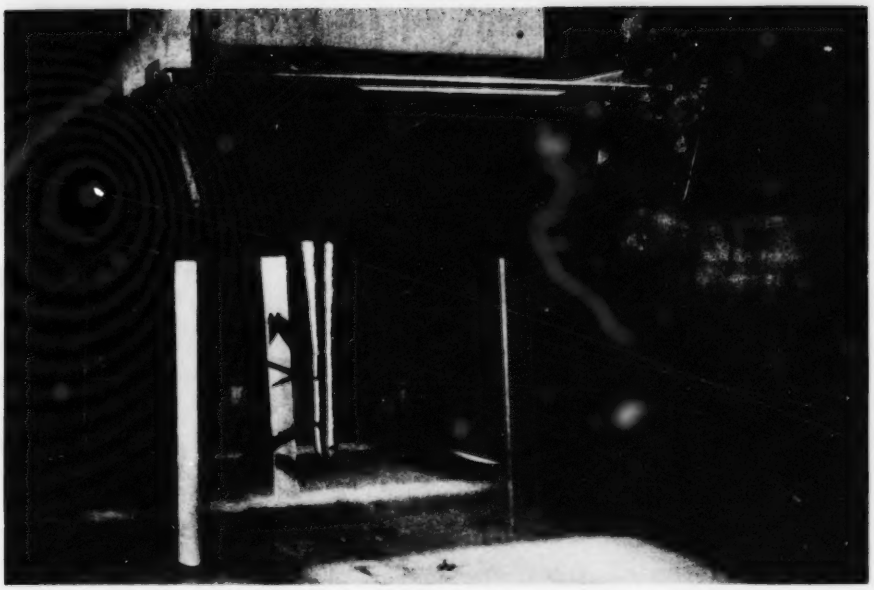


Figure 2 Vertical axis rotor mounted under the barge model

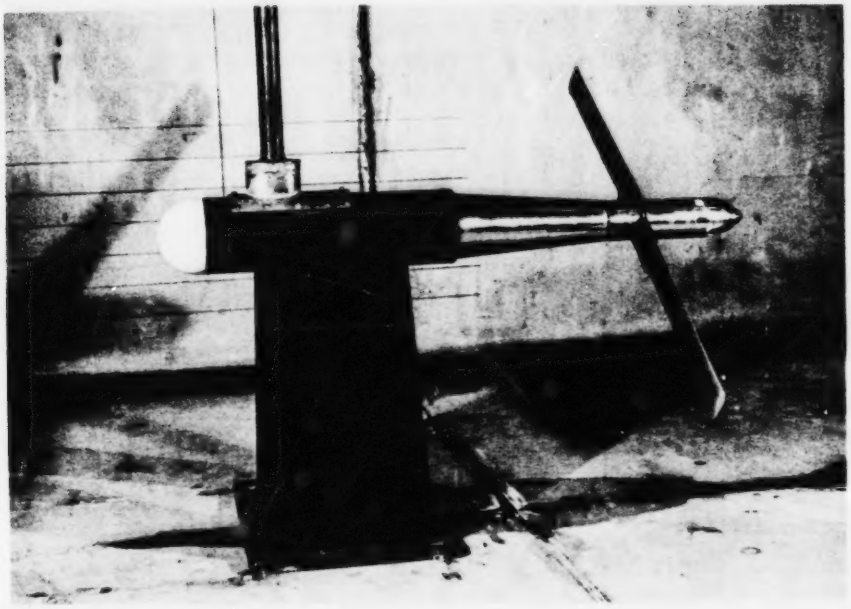


Figure 3 Horizontal axis rotor mounted in the flume during the preliminary tests

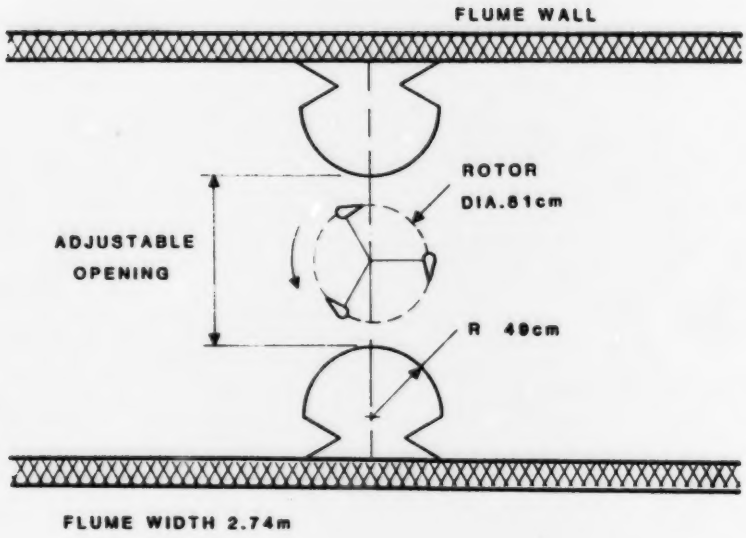


FIGURE 4: DUCT PLAN VIEW

CAISSON

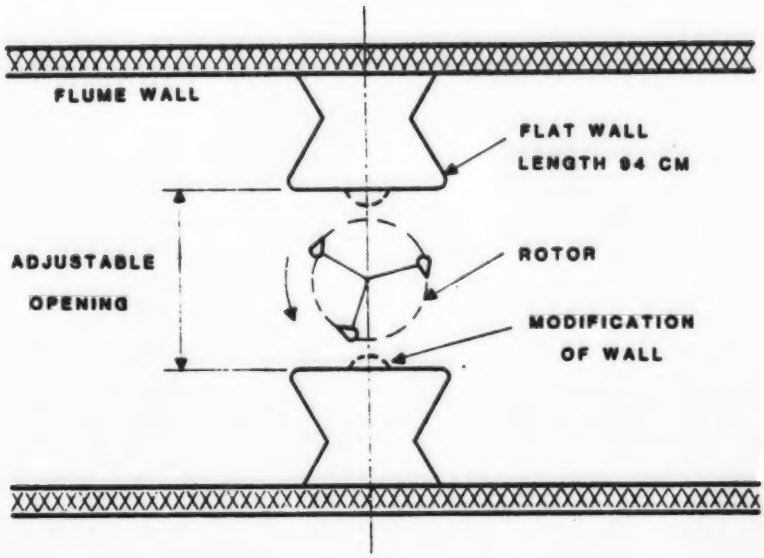


FIGURE 5: FLAT WALL - PLAN VIEW

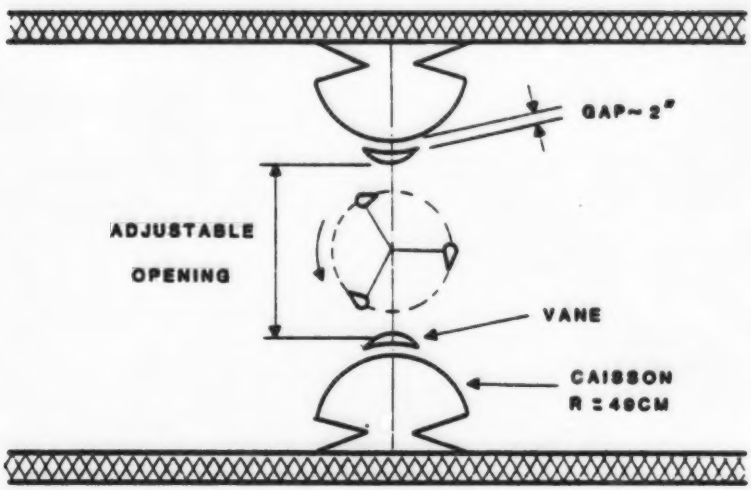


FIGURE 6: DUCT PLAN VIEW

CAISSON AND VANE

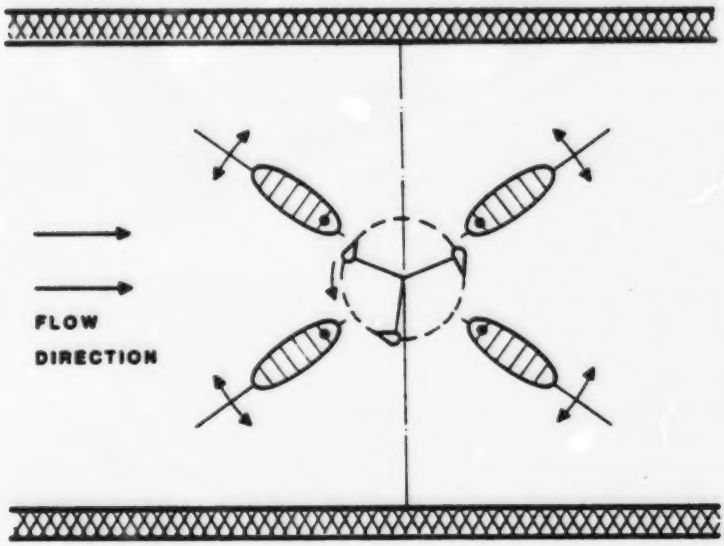


FIGURE 7: PLAN VIEW GUIDE VANES

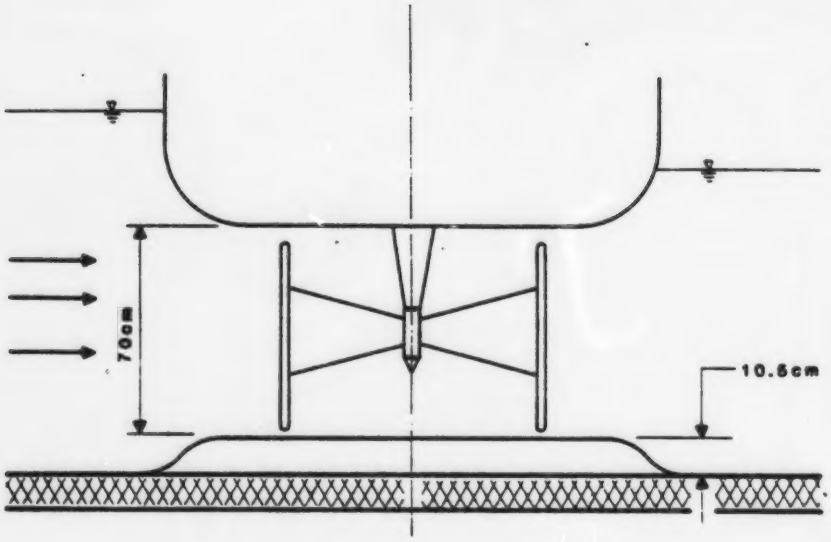


FIGURE 8: TURBINE MODEL BRIDGE STRUCTURE ELEVATION

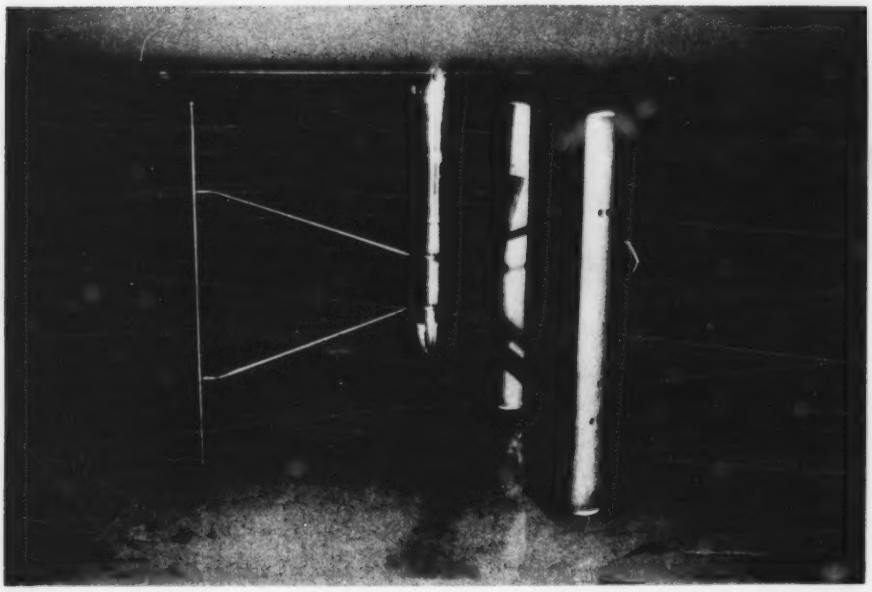


Figure 9 Three bladed rotor in the modified wall duct

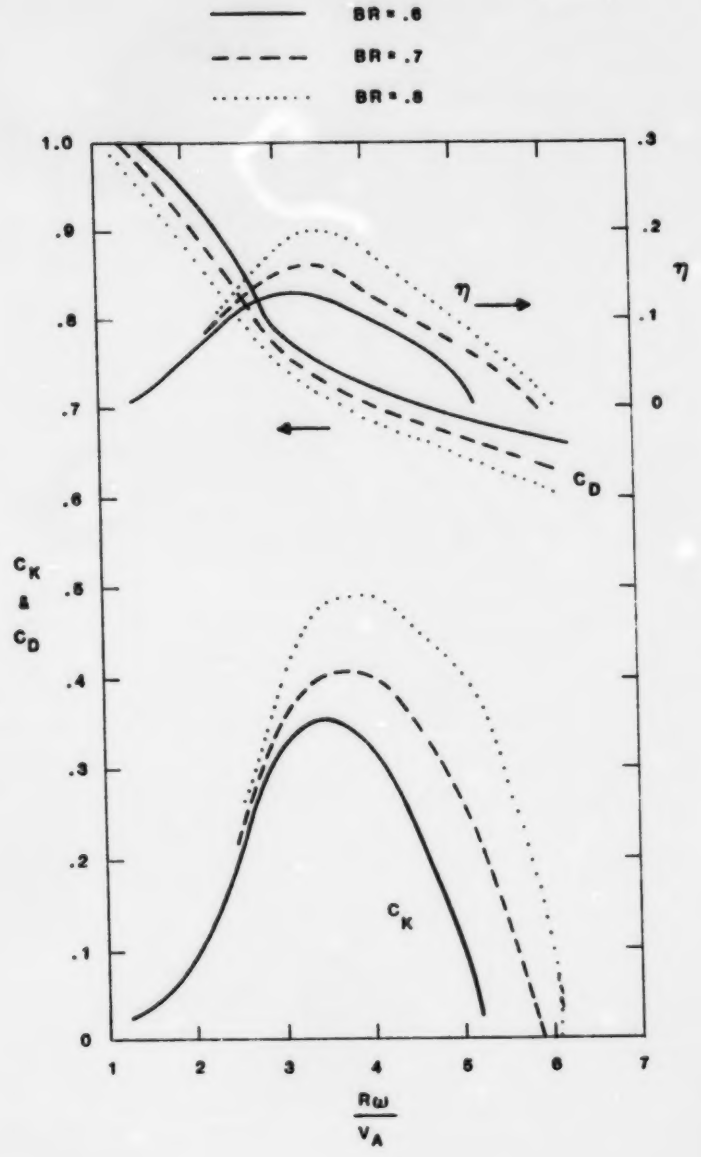


FIGURE 10: EFFECT OF BLOCKAGE RATIO ON DUCTED MODEL STRAIGHT WALL - FROM REF.7

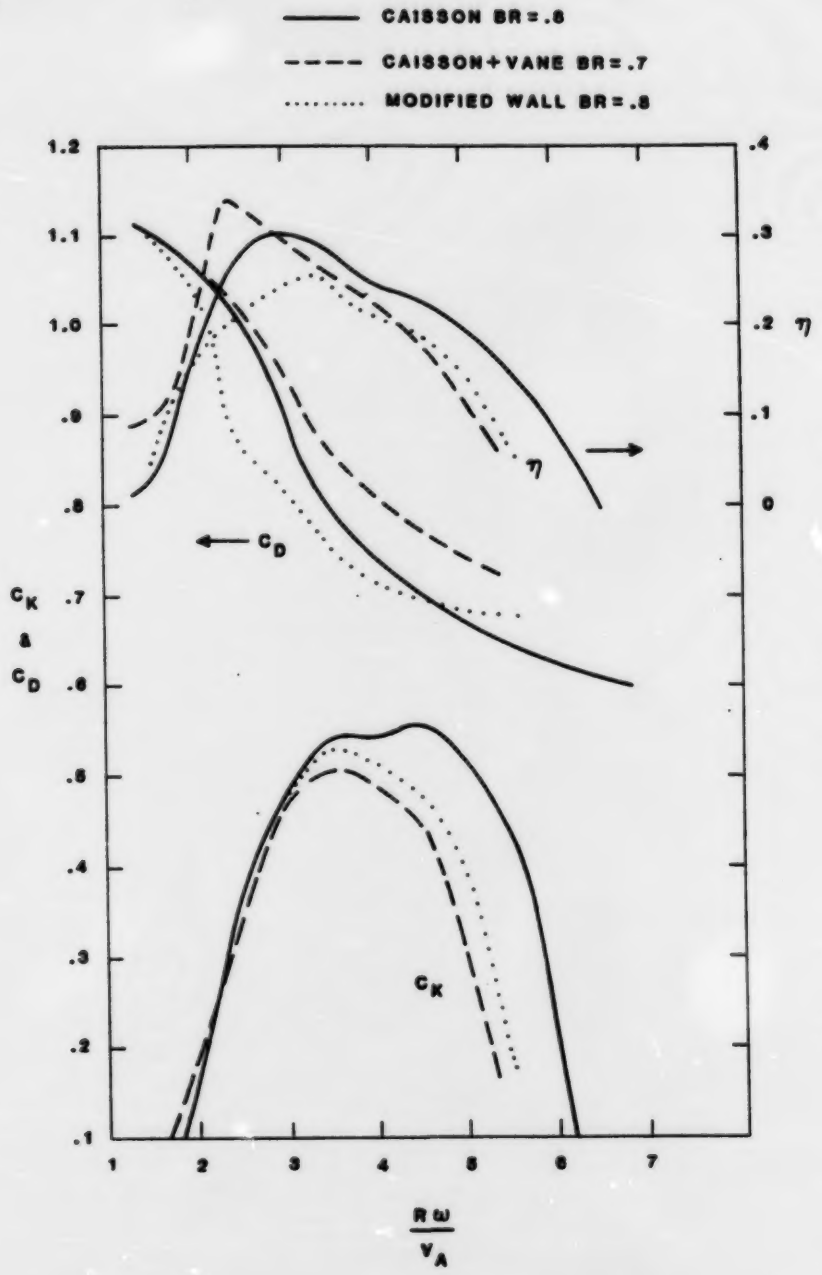


FIGURE 11: COMPARATIVE RESULTS FOR 3 DIFFERENT DUCT SHAPES (MODEL TESTS) FROM REF.7

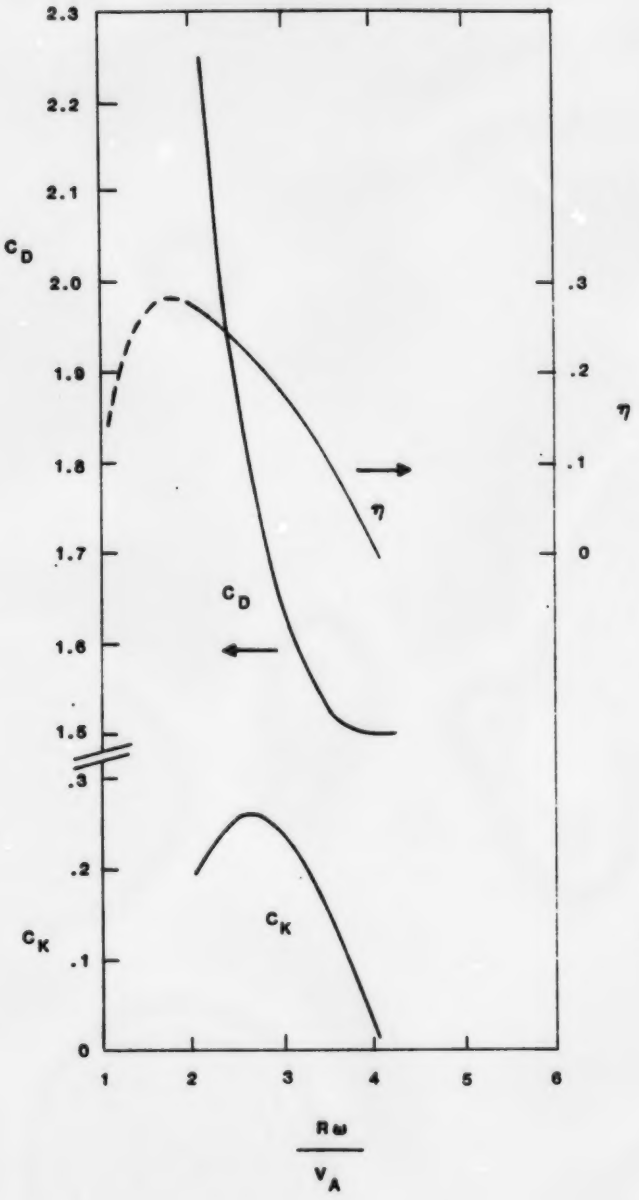


FIGURE 12: CHARACTERISTICS OF MODEL TURBINE.
DUCT REMOVED EXCEPT TOP AND BOTTOM PLATE. BR=.26

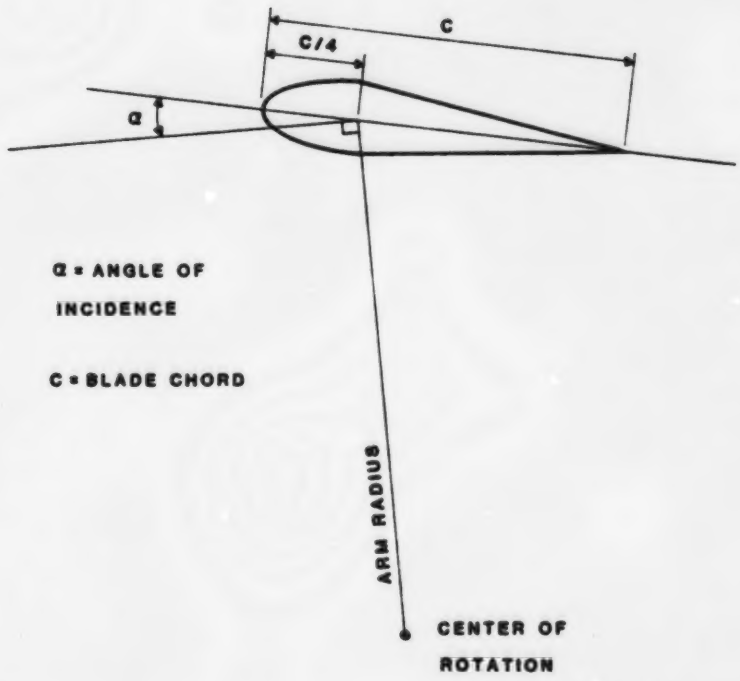


FIGURE 13: BLADE ANGLE OF INCIDENCE

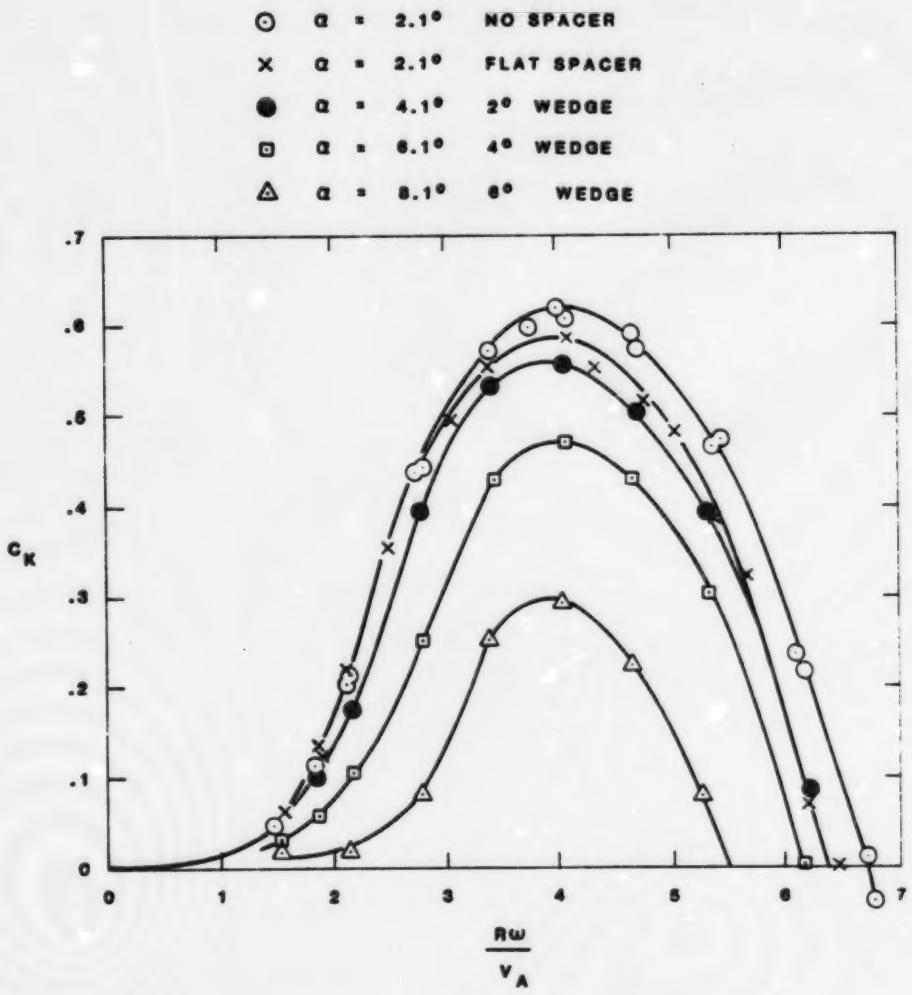


FIGURE 14: EFFECT OF POSITIVE ANGLE OF INCIDENCE

CAISSON DUCT MODEL

BR = .8

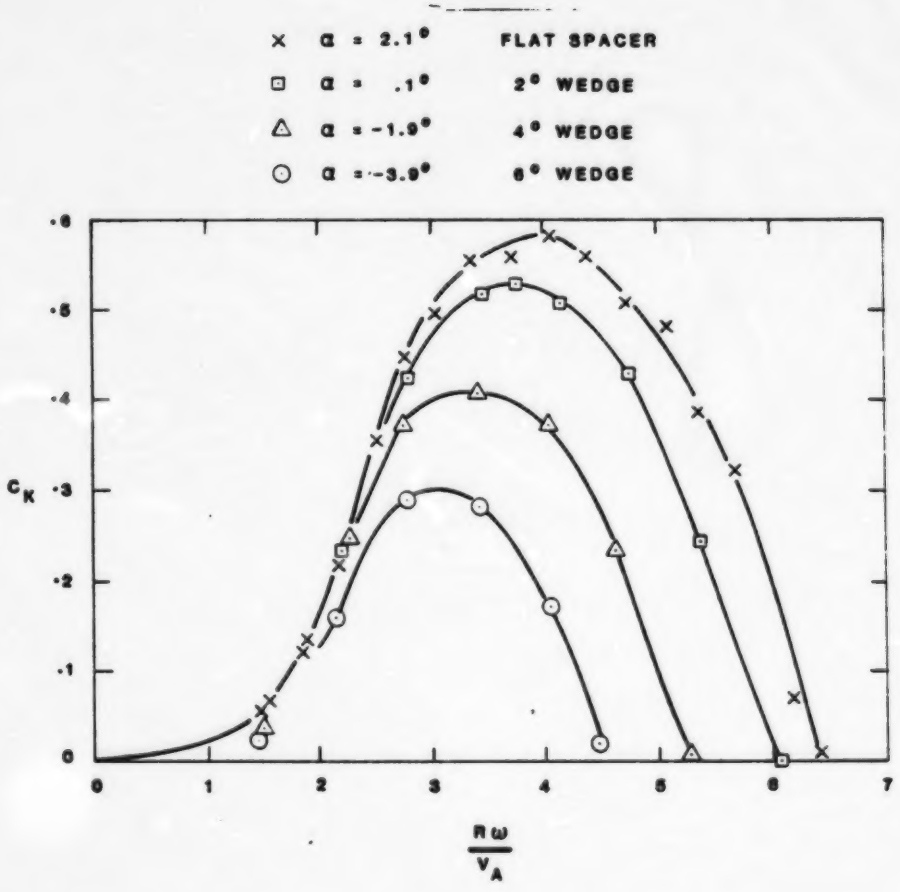


FIGURE 15: EFFECT OF NEGATIVE ANGLE OF INCIDENCE
CAISSON DUCT MODEL
BR = .8

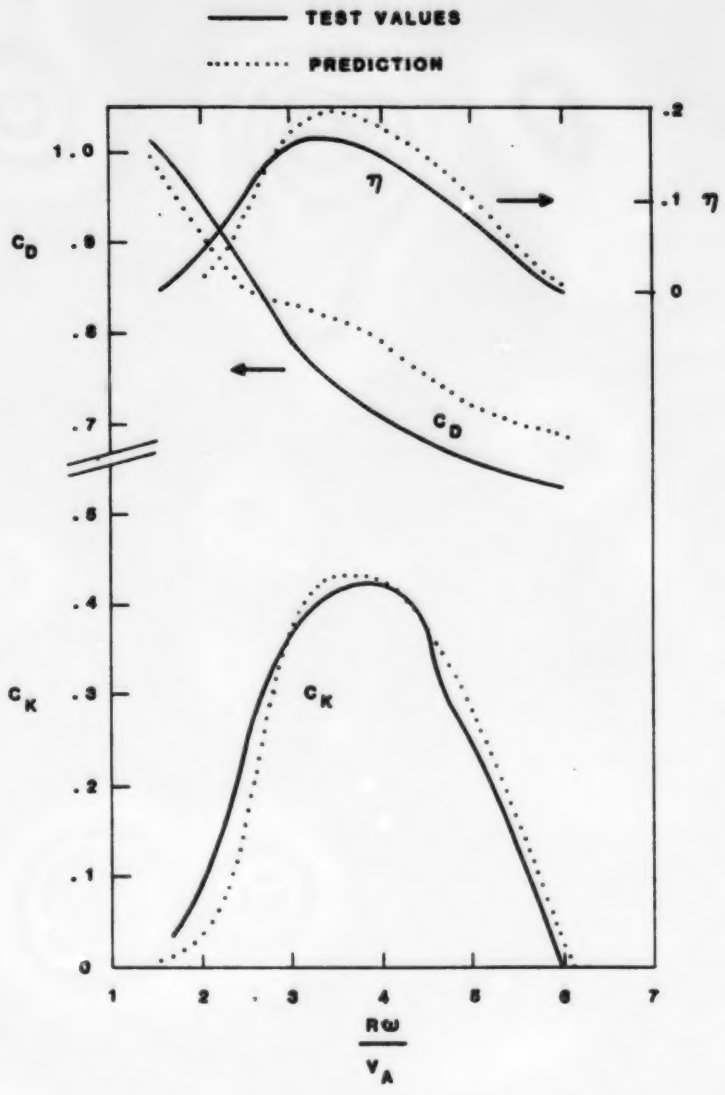


FIGURE 16: COMPARATIVE RESULTS

MODEL TESTS WITH PREDICTION

PROGRAMME - FROM REF.7

STRAIGHT WALLS - BR = .7

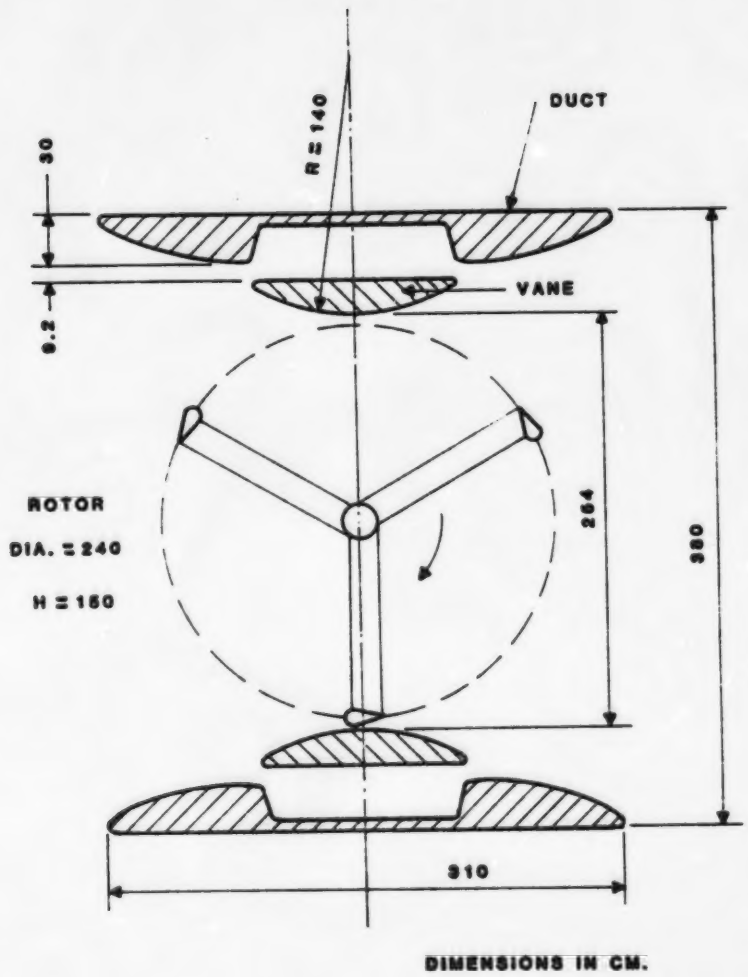
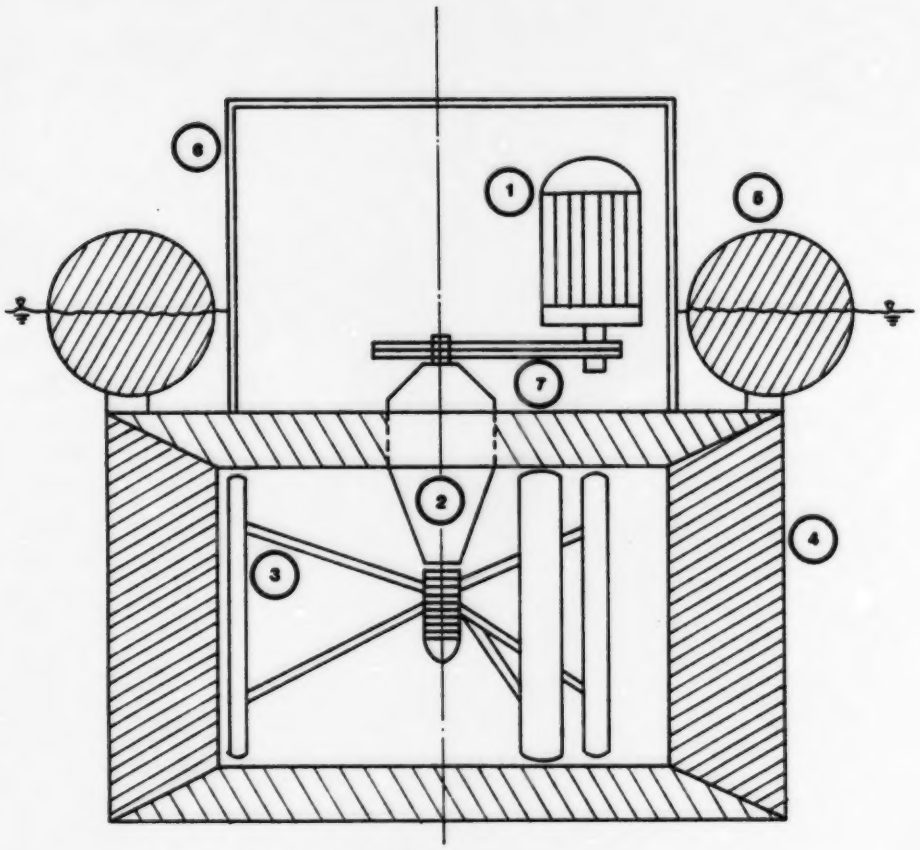


FIGURE 17: CORNWALL PROTOTYPE DUCT ARRANGEMENT



1-INDUCTION GENERATOR
2-SPEED INCREASING GEAR BOX
3-ROTOR
4-DUCT
5-FIBERGLASS PONTOONS
6-WATERTIGHT HOUSING
7-2 BELT DRIVE

FIGURE 18: CORNWALL DUCTED PROTOTYPE
GENERAL ARRANGEMENT

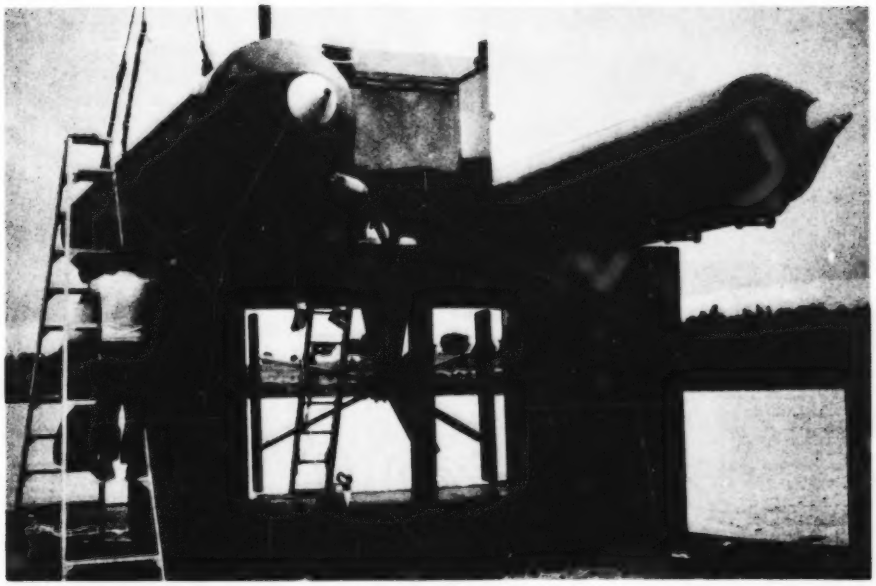


Figure 19 Cornwall prototype during assembly of floatation pontoon

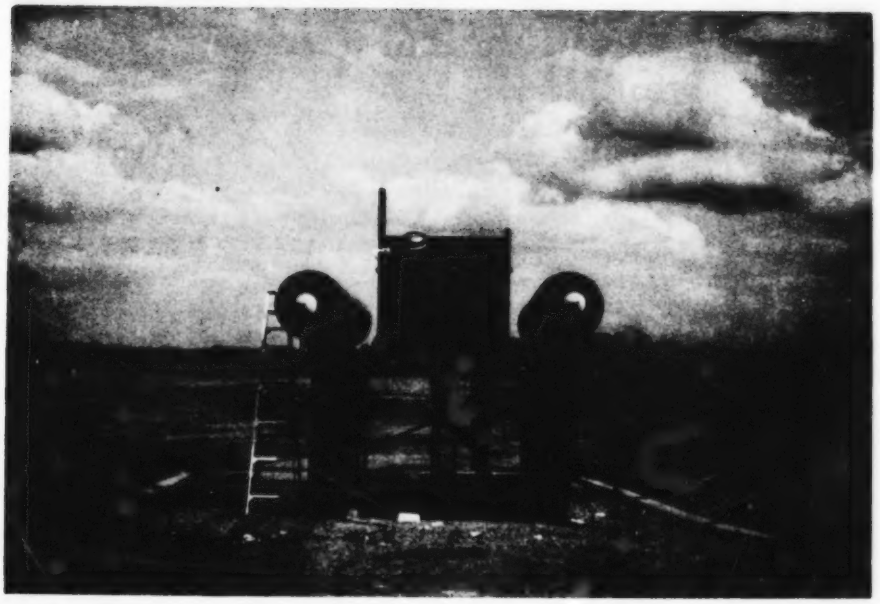


Figure 20 Prototype of ducted turbine prior to launching

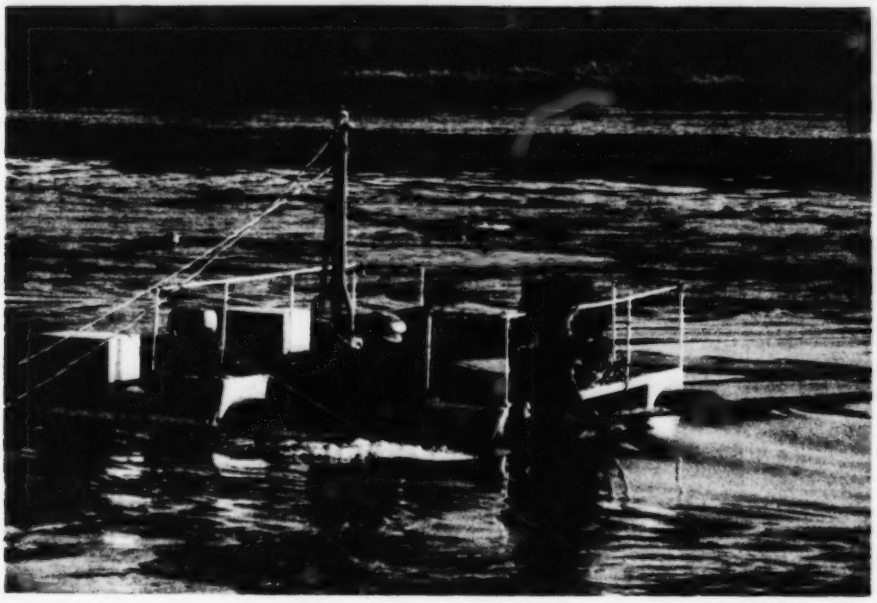


Figure 21 Prototype turbine in location at Cornwall site

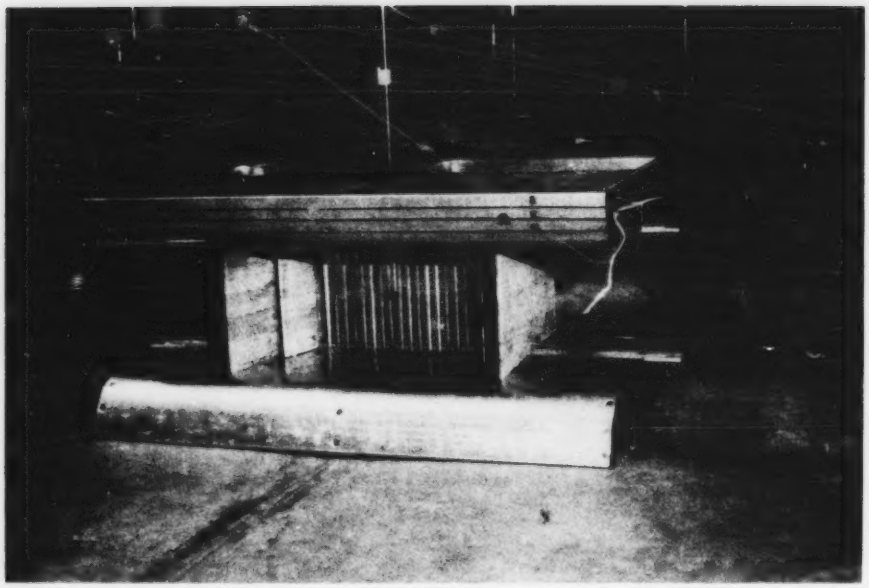


Figure 22 Model of free stream turbine with round bar restriction at the throat of the duct

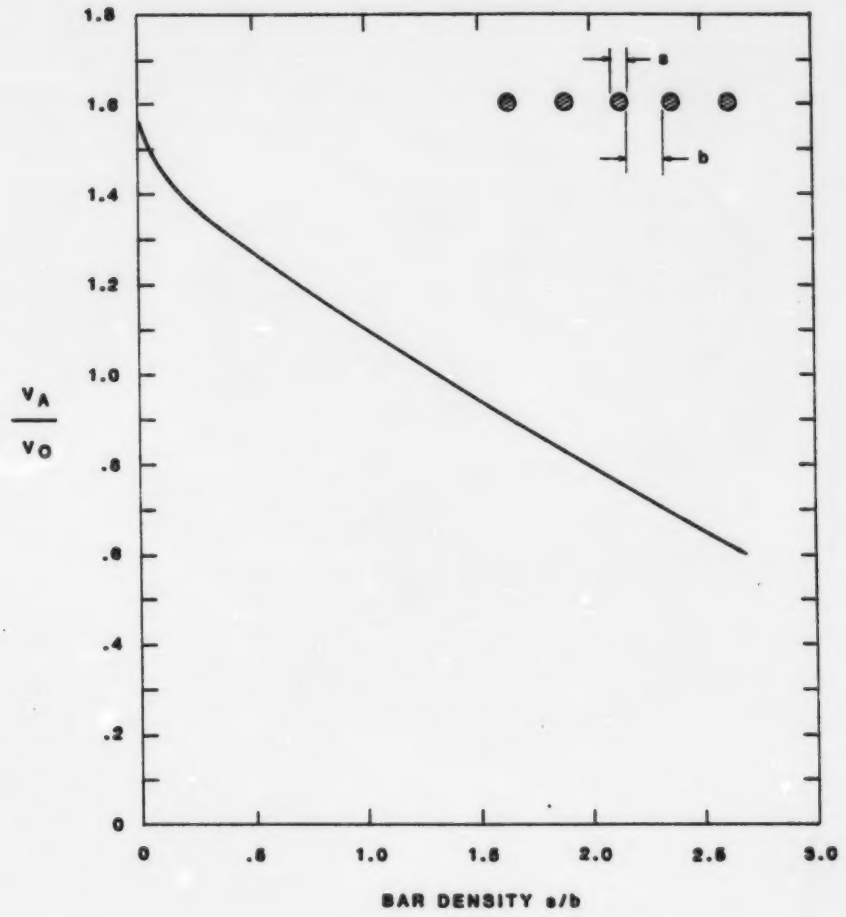


FIGURE 23: MODEL TEST MEASUREMENT OF VELOCITY THROUGH ROUND BARS. SIMULATION OF DUCTED FREE STREAM TURBINE

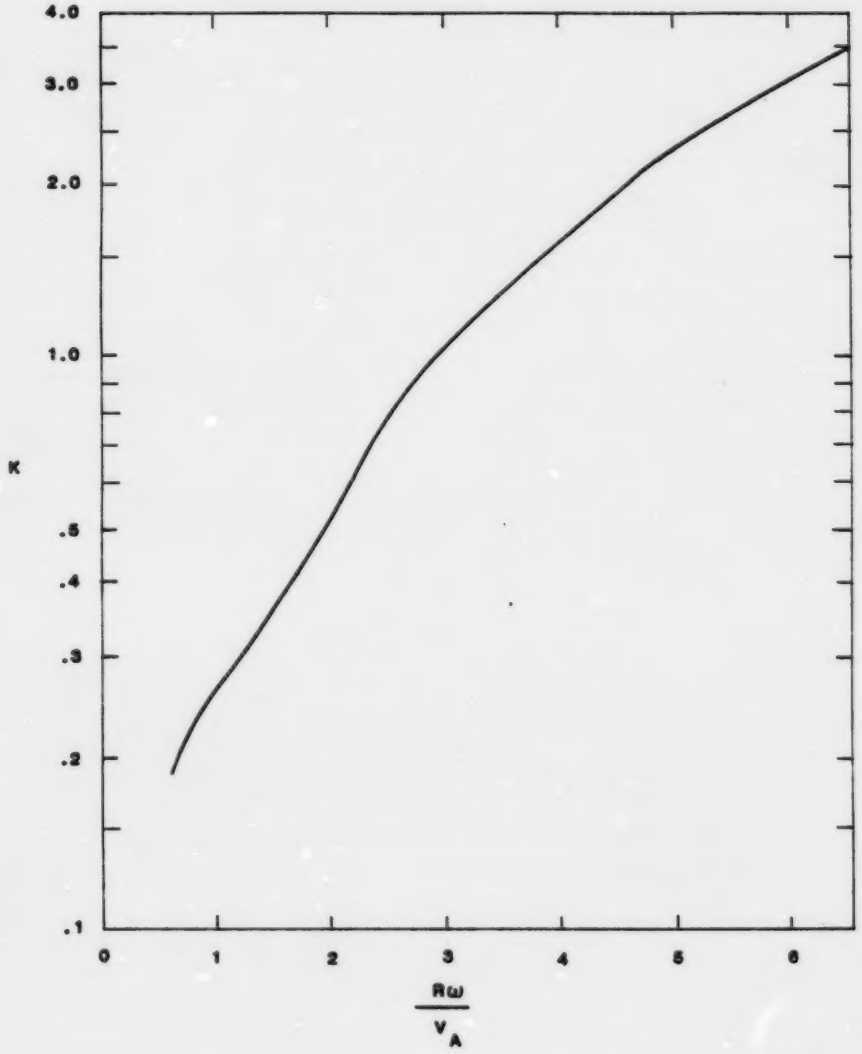


FIGURE 24: PREDICTED ROTOR DRAG COEFFICIENT

FROM REF.7

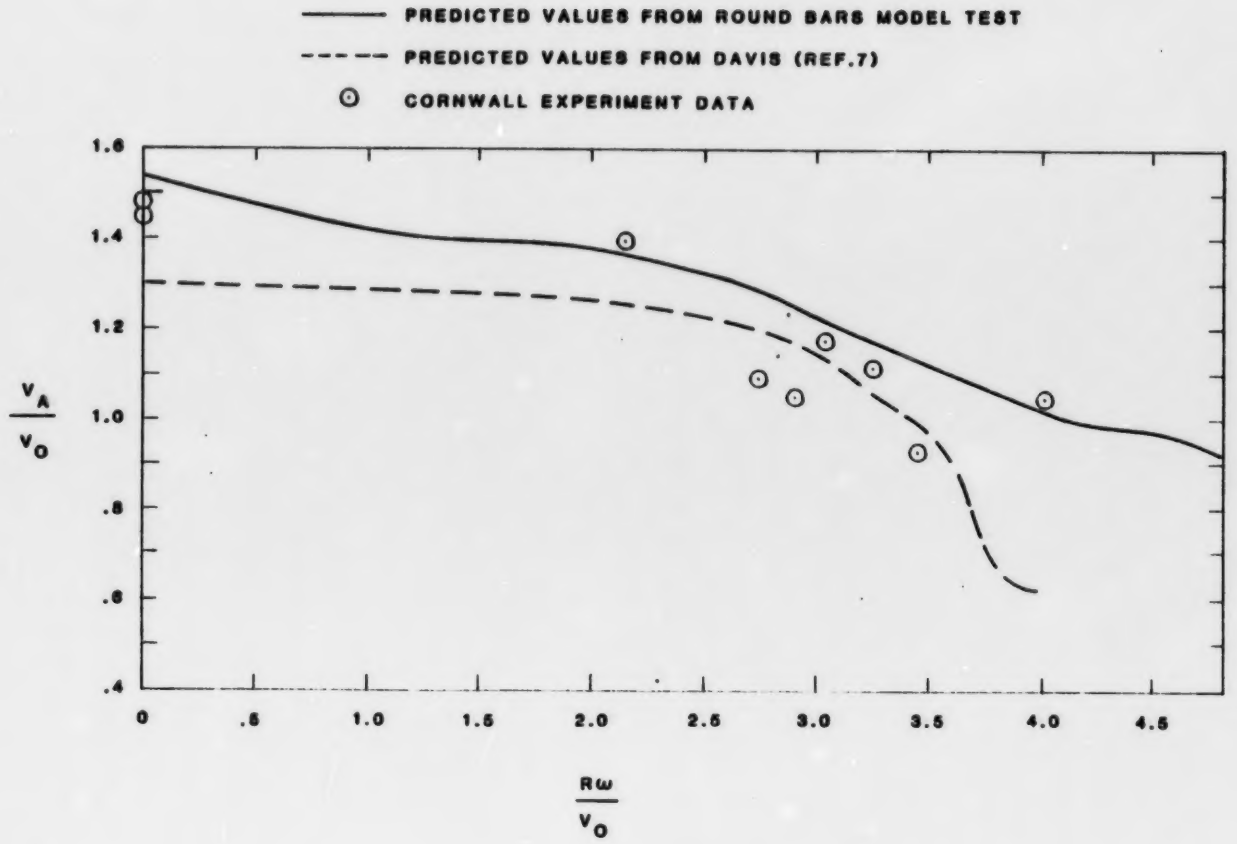


FIGURE 25: VELOCITY THROUGH DUCTED FREE STREAM ROTOR

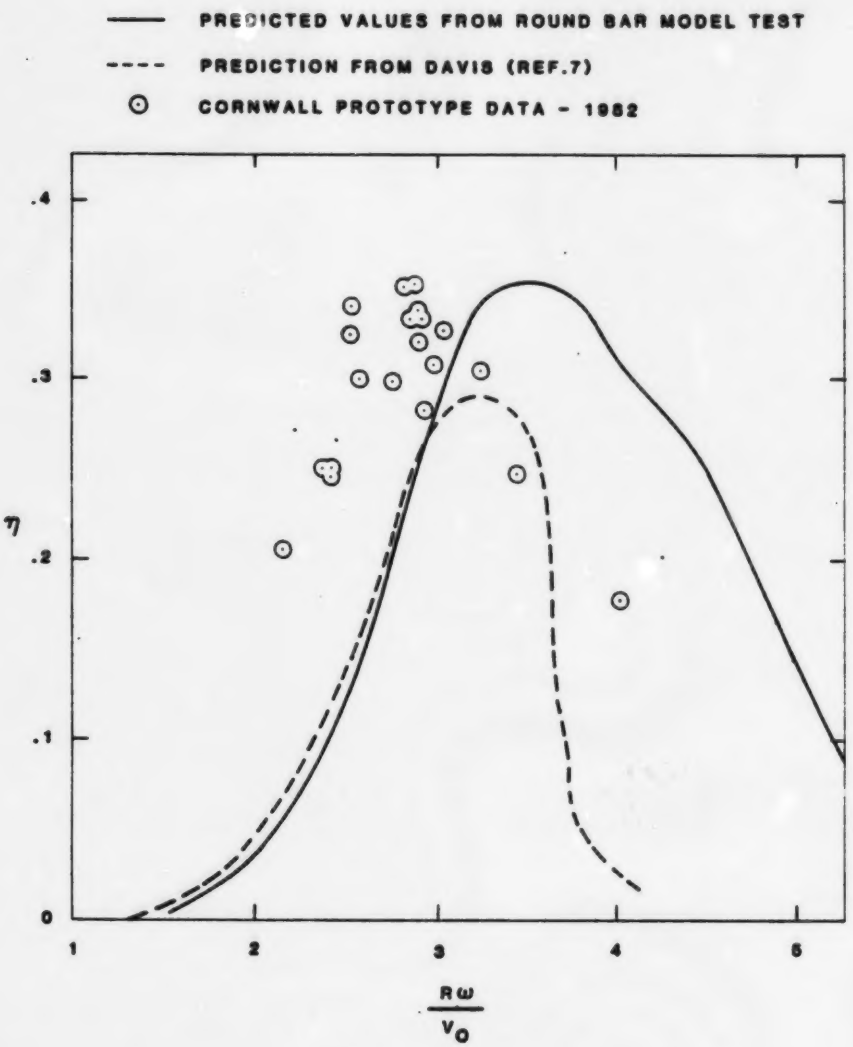


FIGURE 26: EFFICIENCY OF DUCTED FREE STREAM TURBINE

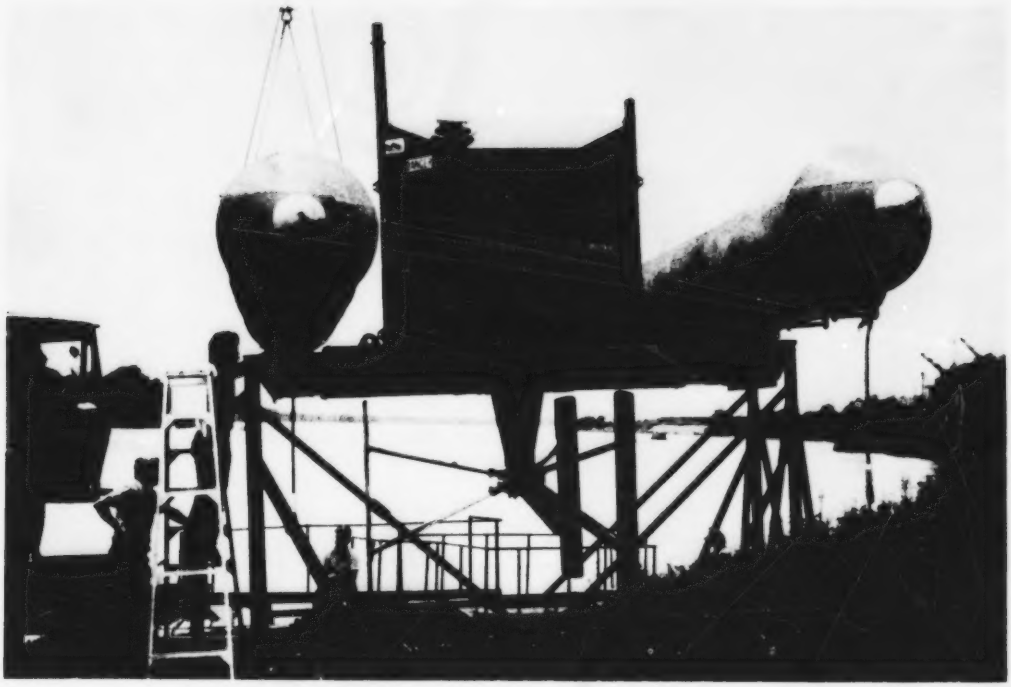


Figure 27 Non ducted free stream prototype with support frame, during pontoon installation

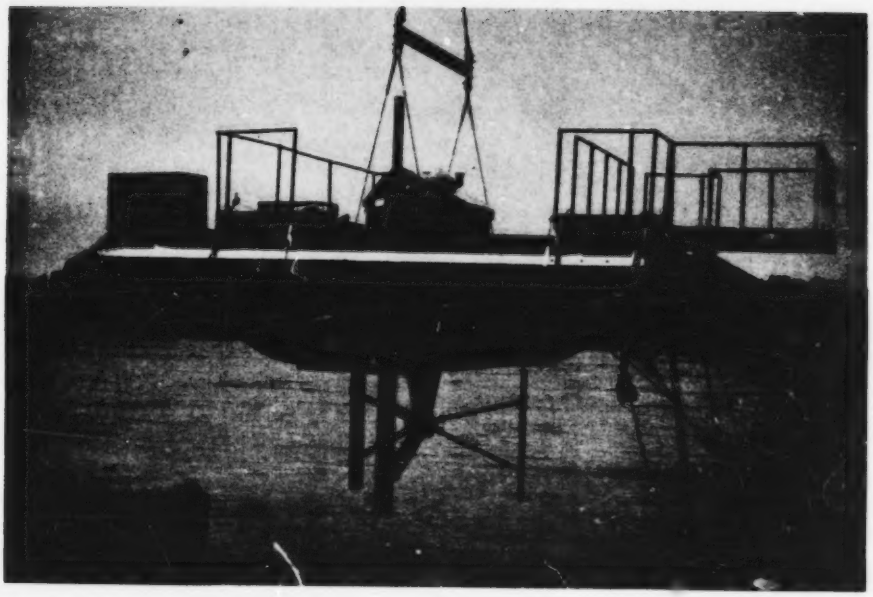


Figure 28 Non ducted free stream prototype during launching at Cornwall, Ontario

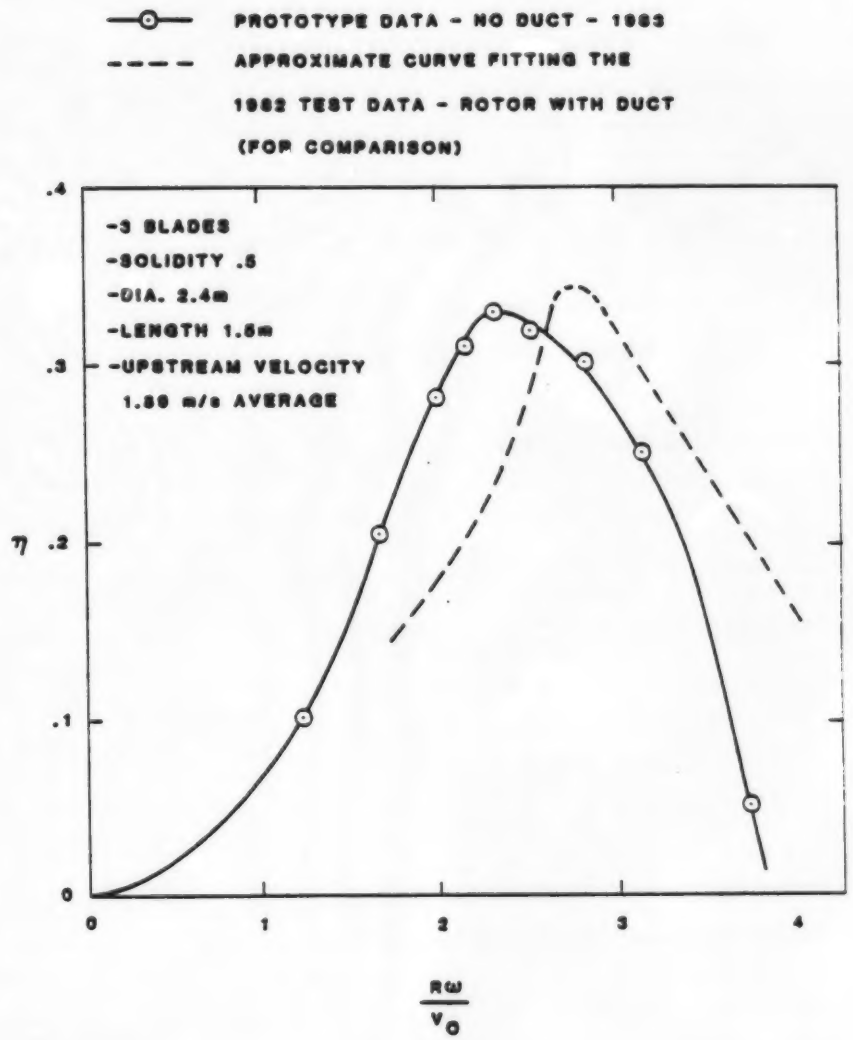
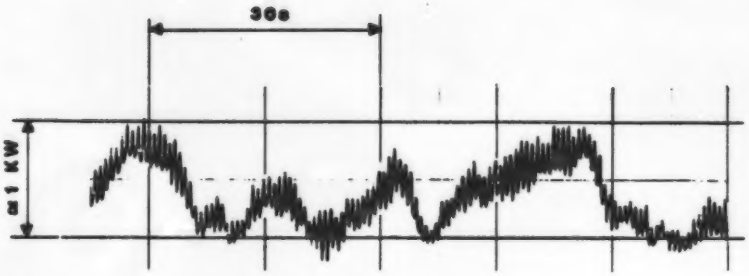
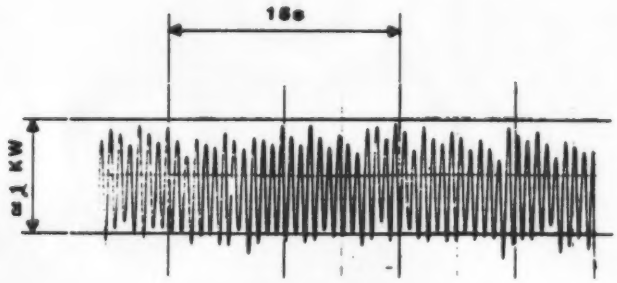


FIGURE 29: EFFICIENCY OF UNDUCTED CORNWALL FREE STREAM TURBINE. PROTOTYPE TEST RESULTS



DUCTED ROTOR 36 RPM

AVERAGE POWER 2.6 KW



NON DUCTED ROTOR 32 RPM

AVERAGE POWER 1.9 KW

FIGURE 30: ELECTRICAL POWER OUTPUT FOR CORNWALL PROTOTYPES

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NOTES

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14

SUMMARY/SOMMAIRE

Model tests of a Darrieus hydraulic turbine installed in a barrage are described. Field tests of a free stream prototype machine are described and test results are compared with theoretical analysis.
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